NON-CONTACTING GAS LUBRICATED FACE SEALS FOR HIGH p · v – VALUES

J. Glienicke, A. Launert, H. Schlums, and B. Kohring
Institute für Maschinenelemente und Fördertechnik
Technische Universität Braunschweig
Braunschweig, Germany

Abstract

The authors inform about recently developed mathematical fundamentals concerning the calculation of non–contacting gas lubricated face seals. They carried out extensive experiments using three different designs at pressures up to 10 MPa and sliding velocities up to 110 m/s. A comparison between the experimental results and the calculations indicates that a stable operation without wear can be ensured in all cases, provided that the materials and geometrical parameters of the seal have been properly chosen.

Introduction

Extreme operating conditions in gas process industry (high pressure, high surface velocity, hazardous and toxic gases) require dry running seals of high reliability with low leakage and minimal wear. Hence self–stabilizing face seals were developed, which are balanced aerostatically and controlled aerodynamically when in operation (refs. 1–9). To achieve a stable clearance between the rotating and the stationary face a self–acting geometry, similar to a narrow aerodynamic thrust bearing, is incorporated into one of the seal faces (ref. 10). As the shaft begins to rotate, the bearing generates a very thin gas film with high stiffness and an opening force, which separates the seal faces. Usually the self–acting geometry is located on the high pressure side.

The operational principle of such a self–stabilizing non–contacting face seal is illustrated in fig. 1: Any deviation in gap width from the position of equilibrium *(index 1)* causes changes in the pressure profile, which determines the opening force. If gap width increases *(2)*, the aerodynamic opening force decreases and vice versa *(3)*. This causes the gap width to automatically return to its equilibrium position. At present three self–acting surface patterns as shown in fig. 2 are used in industrial applications.

There has been a growing interest in these seals as components for process industry and other ranges (turbo compressors, expansion and cooling turbines etc.) in the last 15 years, but nevertheless there have been a certain lack of design fundamentals until recently. Therefore research projects were performed to obtain reliable, experimentally verified fundamentals for the design and calculation of gas lubricated face seals used at high p·v – values.

Theoretical Fundamentals

The isothermal compressible fluid flow in a sealing/lubricating gap is described by
- the Navier-Stokes-equation,
- the equation of continuity and
- the equation of state (ideal gas)

with the boundary conditions and the simplifications commonly used in aerodynamic lubrication theory being applied. The generalized Reynolds equation results from this system of equations and is solved numerically to determine the pressure distribution in the sealing gap as well as the static and dynamic characteristics of gas lubricated face seals (ref. 11, 12).

Turbulence in the lubricating film may occure at very high surface velocities and high pressure differences. The effect of turbulence is represented by the two turbulence correction factors K_Θ and K_R, which are based on empirically derived turbulence models developed by Ng and Elrod as well as Constantinescu (ref. 13).

The generalized Reynolds differential equation is:

$$\frac{1}{r}\frac{\partial}{\partial\Theta}\left(\frac{\rho h^3}{\eta K_\Theta}\cdot\frac{\partial p}{\partial\Theta}\right)+\frac{\partial}{\partial r}\left(\frac{r\rho h^3}{\eta K_R}\cdot\frac{\partial p}{\partial r}\right)=6\omega r\frac{\partial(\rho h)}{\partial\Theta}+12r\frac{\partial(\rho h)}{\partial t} \qquad (1)$$

with: p – pressure ω – angular velocity
         h – gap width Θ – angular cordinate
         η – dynamic viscosity r – radial coordinate
         ρ – density t – time.

In solving the Reynolds equation numerically the mean film thickness and the pressures at the boundaries are held constant in each iteration. The pressure distribution $p(\Theta, r, t)$ is calculated whereby the stationary and transient contributions of the solution are computed separately. A conservative Finite-Difference-Method is used because of the discontinuities in the gap between the stationary and the rotating seal ring.

The gap opening force given by the integration of the pressure distribution is iterated, varying film thickness, until the equilibrium of opening and closing force is achieved (ref. 14). The friction force F_R is calculated by integrating the wall shear stress with respect to the seal face. The leakage (mass flow $\dot{m}$) results from the exit velocity times exit area and density. The linearized stiffness and damping coefficients (c_a, d_a) are determined by applying the method of small perturbation to the Reynolds equation.

Additionally the commercial FEM-program ANSYS (ref. 15) is coupled to this flow analysis program to estimate the influence of seal deformations on the characteristics.

For the design of gas lubricated face seals the effect of pressure and speed on leakage and gap width is of great importance. In the following the calculated characteristics of a gas lubricated face seal with spiral grooves and high pressure on the outer diameter are

presented. In this example it is assumed that the gas flow in the sealing gap is laminar and that the seal faces are parallel. The process medium is air.

The leakage of this spiral groove face seal shows nearly a linear increase with operating pressure and speed, fig. 3a. As shown in fig. 3b, the sealing gap decreases slightly with increasing pressure and increases with growing speed. At maximum operating pressure, $p_{op,max} = 10$ MPa, and max. speed, $n_{max} = 22\,000$ rpm, the gap width is $h = 2.5$ μm. The resulting leakage mass flow of $\dot{m} = 1.45$ g/s is comparatively low. The seal is closed at low pressure difference and $n = 0$.

During operation an angular misalignment of the rotating seal ring and an axial shaft movement are always present. To provide good dynamic tracking ability between the stationary and the misaligned rotating ring the flexible support of the stationary ring in the housing allows angular and axial movements. To ensure stable operation it is essential, that the axial and angular stiffnesses of the gas film are very high and that the damping of the film is positive. With the chosen spiral groove design these requirements are met as shown in fig. 4.

Experimental Investigations

To verify the reliability of the developed design fundamentals series of comparative experiments using different complete gas lubricated face seals were carried out. The tests were performed with three seal designs at operating pressures p_{op} up to 10 MPa and sliding velocities v up to 110 m/s.

In fig. 5 the cross-section of the test rig is shown. The shaft is supported in ball bearings additionally equipped with a squeeze film damper on the side of the sealing unit. The test seals are arranged symmetrically to compensate the axial pressure force. Various sizes, designs and materials were tested for the seal rings, as well as different pattern for the self–acting structure and different balance ratios.

Test seal A (John Crane) is patterned with spiral grooves. The stationary face is manufactured from carbon – graphite and the rotating seal ring (outer diameter 148 mm) from tungsten carbide. The measured total leakage volume flow $\dot{V}_{tot}$ (fig. 6) increases very strong with speed and operating pressure and coincides quite well with the calculated values. The measured total power consumption P_{tot} (fig. 7) varies progressively with speed n and about linearly with pressure p; at a maximum speed of $n_{max} = 16\,000$ rpm and a maximum pressure of $p_{max} = 8$ MPa it reaches a value of $P_{tot} = 4$ kW. This high power consumption is due to highly turbulent flow at the surfaces of the rotating elements inside the pressurized chamber.

Test seal B (Pacific Wietz) uses a self–acting structure of symmetrical T – grooves. The material of the stationary ring is silicon carbide filled with graphite and that of the rotating ring is tungsten carbide (outer diameter 98 mm). The leakage characteristics (mass flow $\dot{m}$) of this seal (fig. 8a) are similar to those of seal A above, but the influence of speed is very small. To measure gap width during operation the stationary and rotating ring were modified. At the inner diameter of the stationary ring (atmospheric pressure side) two capacitance probes were mounted. The measured gap width h (fig. 8b) ranges from 2μm to 3.5μm across the entire operating range. The intersection of the measured lines is caused by deformations of the stationary ring. The results of calculation, which also takes into account the deformation of the sealing rings, agree quite well with the measurements.

The self–acting structure of test seal C (Feodor Burgmann) consists of circular V – grooves, which reach wedge shaped inwards (grooves depth is not constant). Here again the stationary ring is made of carbon and the rotating ring of silicon carbide (outer diameter 118 mm). As illustrated in fig. 9a the speed n very strongly influences the leakage $\dot{V}$ due to the high aerodynamic forces generated by the V – grooves. The effect of the operating pressure on leakage is at first very strong for low pressures (p_{op} <4 MPa) and decreases with higher pressures. The film thickness between the seal faces was measured by a capacitive probe. One of its electrodes was sputtered on the sealing face of the stationary ring (Al_2O_3). With no pressure difference across the seal it runs like a gas lubricated aerodynamic thrust bearing. For increasing operational pressure the film thickness decreases, fig. 9b. In operation the gap width h usually varies between the limits 2μm and 6μm.

Summary

In the investigated gas lubricated face seals a very narrow non–contacting sealing gap is realized, which is self–stabilizing. The theoretical fundamentals for a reliable design and optimization of the static and dynamic characteristics of these seals have been developed and are presented here, as well as in other recent publications (refs. 14,16). The reliability of the method for calculating the characteristics of non–contacting gas lubricated face seals at high p·v – values has been verified by extensive comparative tests performed for three seal designs at operation pressures up to 10 MPa and sliding velocities up to 110 m/s. If the materials and geometric parameters of the sealing rings and the self–stabilizing patterns on the seal surface have been chosen properly, a non–contacting, stable and reliable operation without wear can be achieved.

Remarks

Some important problems of gas lubricated seals are not discussed here, for example: choked flow, energy dissipation, entrance and exit losses etc. These topics are discussed in other publications or else will be investigated in future projects.

Acknowledgements

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Figures
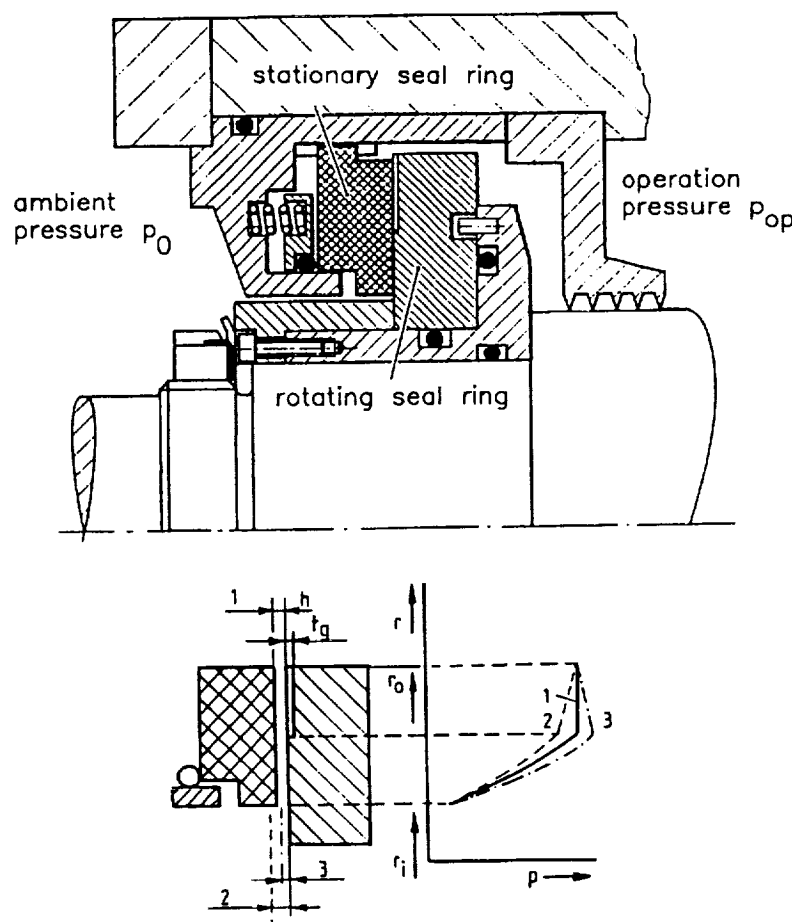
Figure 1: Design and operational principle of gas lubricated face seals

spiral grooves   T–grooves   V–grooves
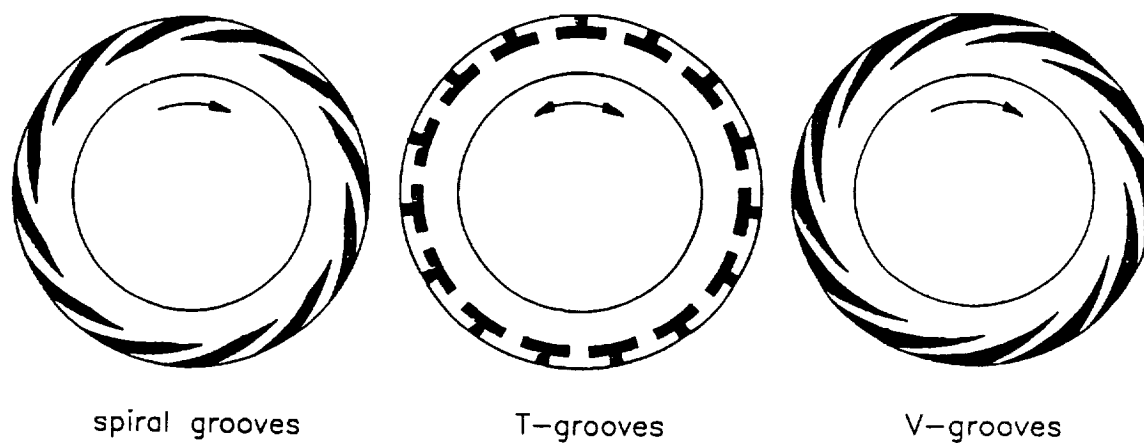
Figure 2: Self–acting geometries of gas lubricated face seals
(⟶ Sealing ring's direction of rotation)
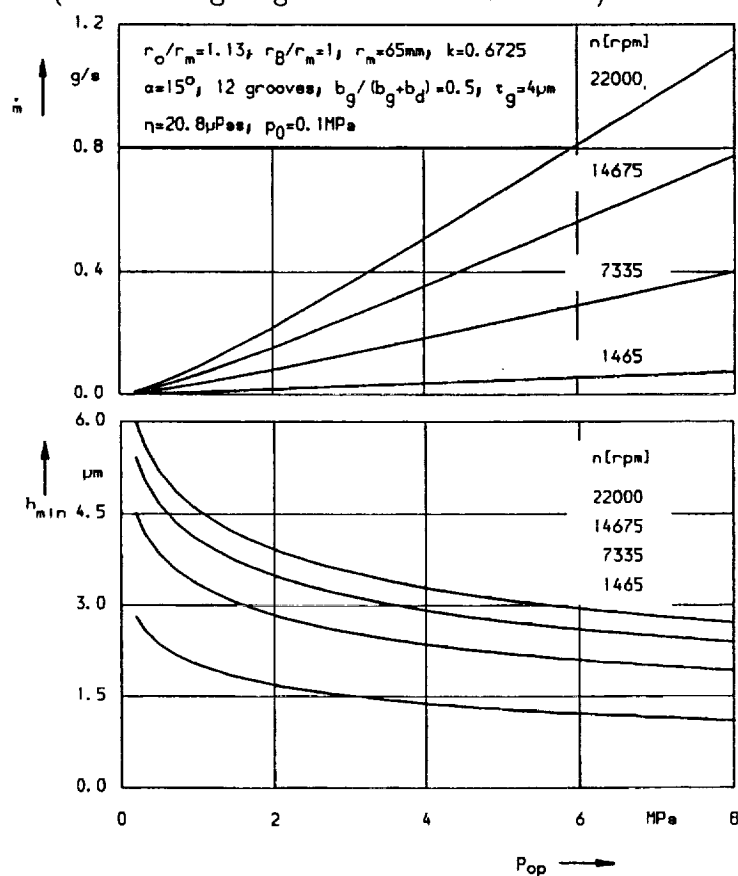
Figure 3: Calculated leakage mass flow $\dot{m}(p_{op}, n)$ and film thickness $h(p_{op}, n)$ of a spiral groove face seal
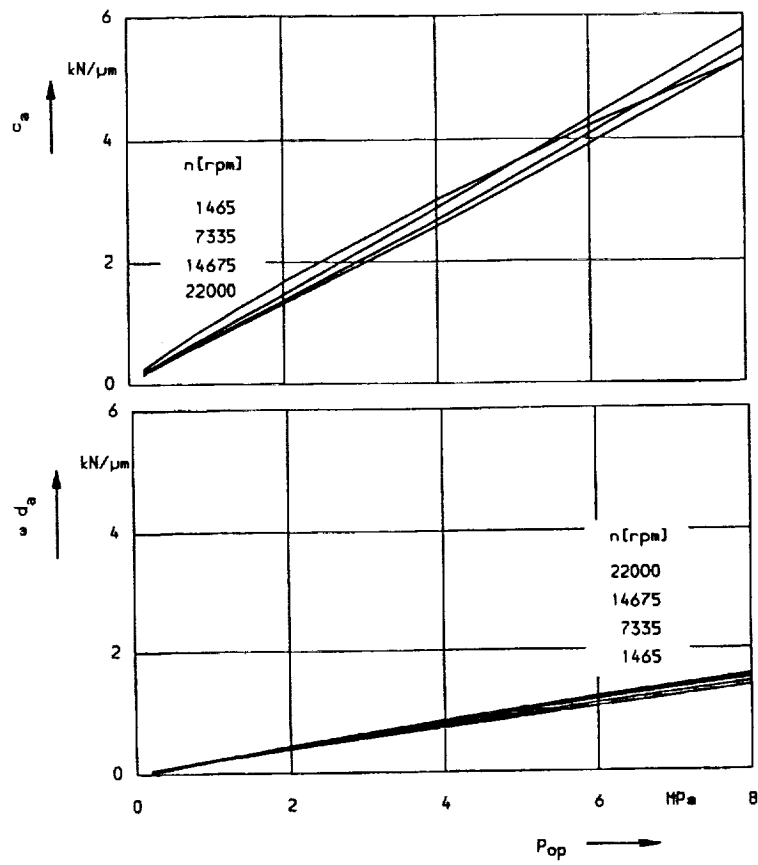
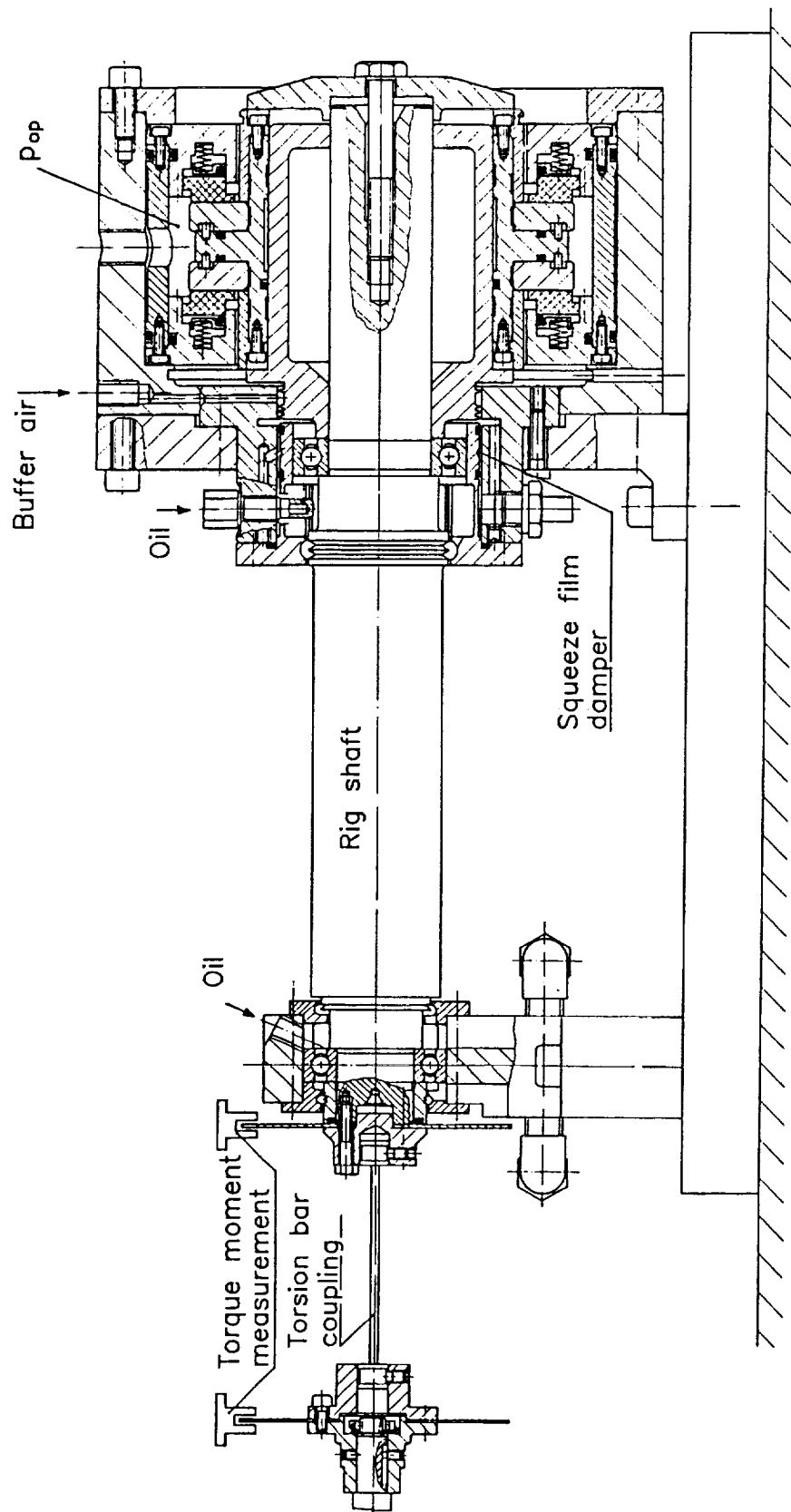Figure 4: Calculated axial stiffness $c_a(p_{op}, n)$ and damping $d_a(p_{op}, n)$ (spiral groove face seal of fig. 3)

Figure 5: Test rig for gas lubricated face seals
($P_{max} = 7$ kW, $n = 0...30000$ rpm, $p_{op,max} = 10$ MPa)
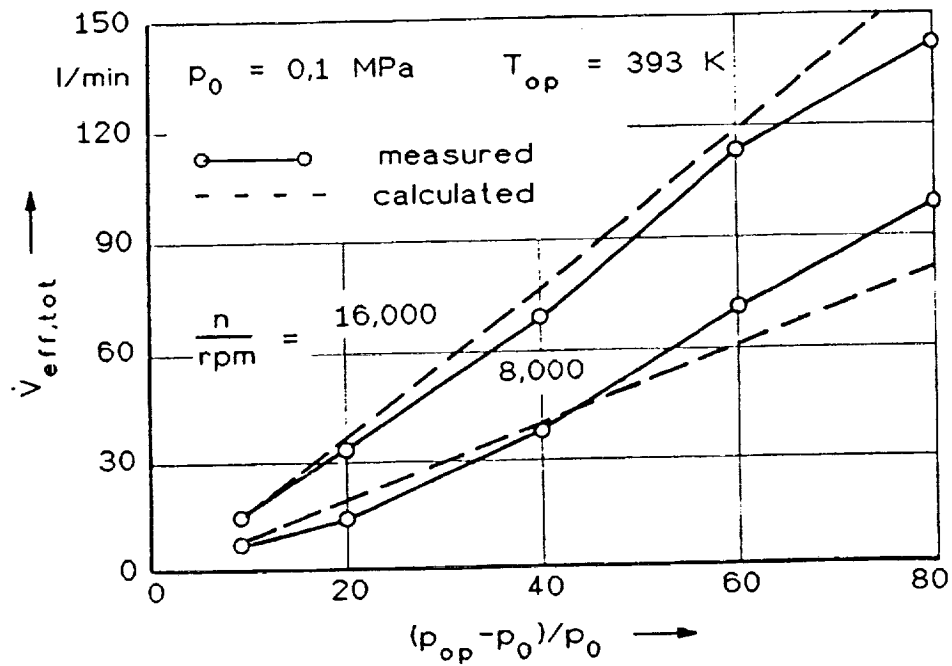
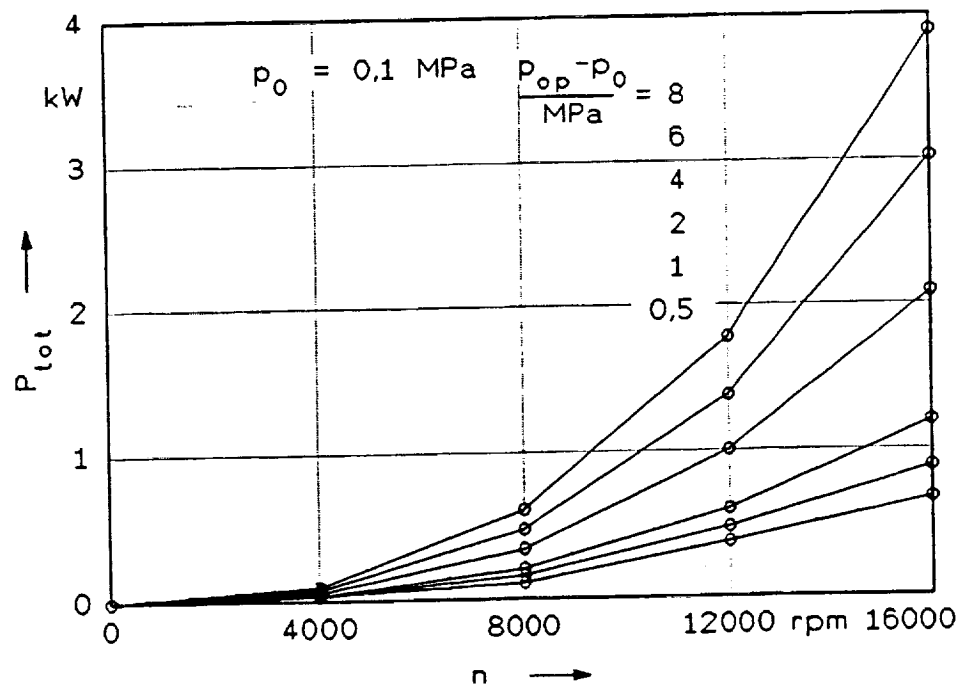Figure 6: Total leakage volume flow $\dot{V}_{tot}(p_{op}, n)$ of test seal A

Figure 7: Measured power consumption $P_{tot}(n, \Delta p)$ of test seal A
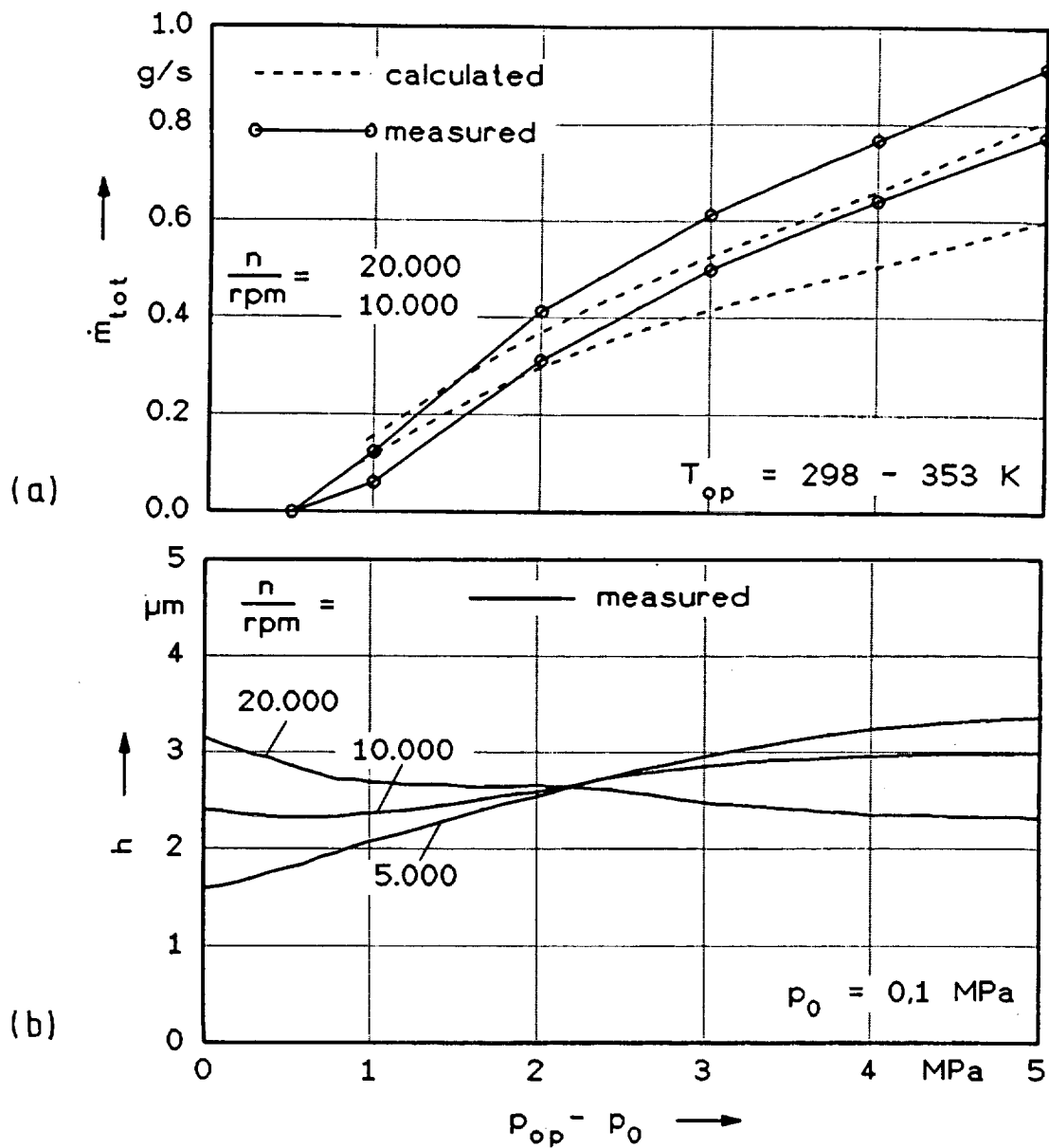
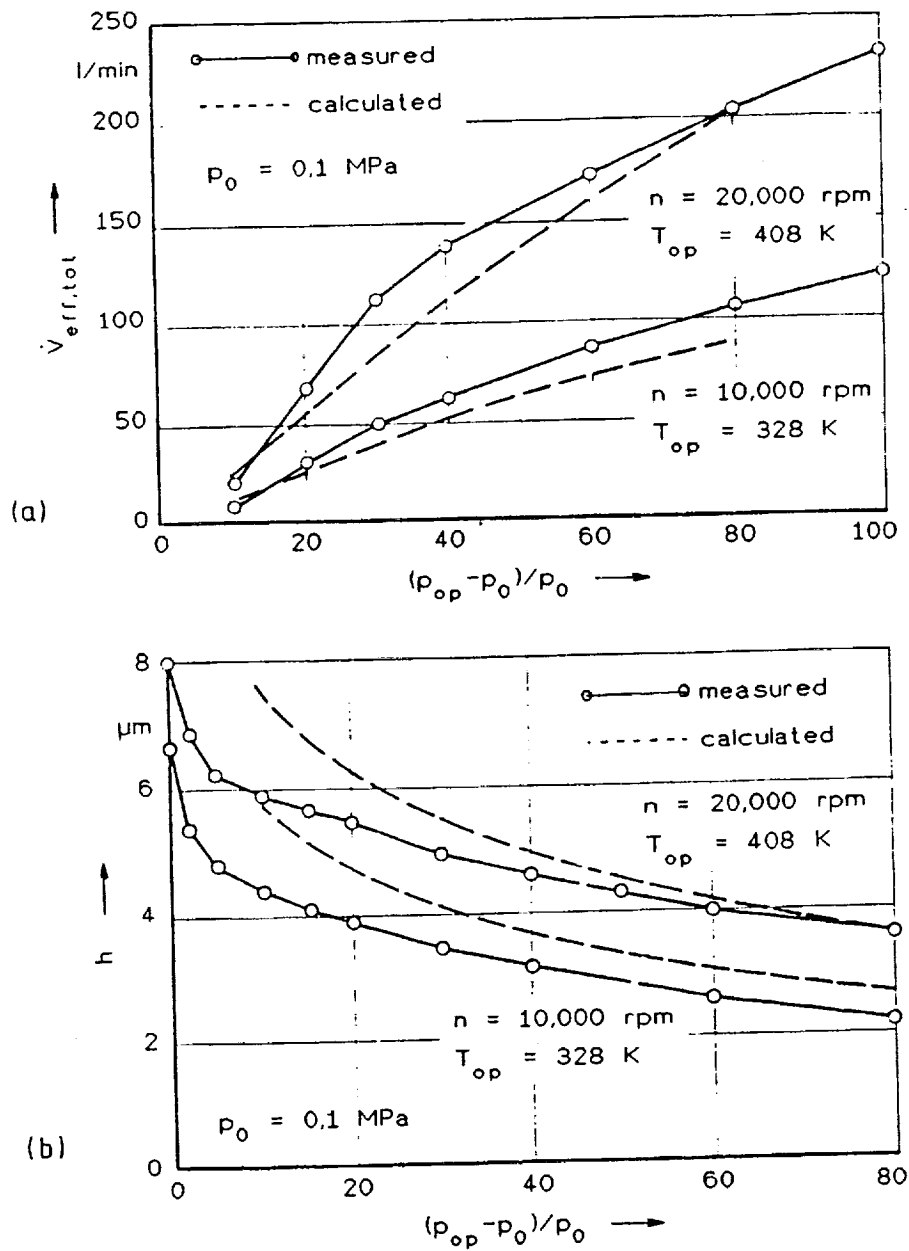Figure 8: Leakage $\dot{m}_{tot}(\Delta p, n)$ (a) and gap width $h(\Delta p, n)$ (b) of test seal B

Figure 9: Leakage $\dot{V}_{tot}(p_{op}, n)$ (a) and gap width $h(p_{op}, n)$ (b) of test seal C

REPORT DOCUMENTATION PAGE

Form Approved
OMB No. 0704-0188

1. AGENCY USE ONLY *(Leave blank)*	2. REPORT DATE	3. REPORT TYPE AND DATES COVERED
	July 1994	Conference Publication

4. TITLE AND SUBTITLE

Seals Flow Code Development-93

5. FUNDING NUMBERS

WU-590-21-11

6. AUTHOR(S)

Anita D. Liang and Robert C. Hendricks, compilers

7. PERFORMING ORGANIZATION NAME(S) AND ADDRESS(ES)

National Aeronautics and Space Administration
Lewis Research Center
Cleveland, Ohio 44135-3191

8. PERFORMING ORGANIZATION REPORT NUMBER

E-8643

9. SPONSORING/MONITORING AGENCY NAME(S) AND ADDRESS(ES)

National Aeronautics and Space Administration
Washington, D.C. 20546-0001

10. SPONSORING/MONITORING AGENCY REPORT NUMBER

NASA CP-10136

11. SUPPLEMENTARY NOTES

Proceedings of a workshop held at NASA Lewis Research Center, Cleveland, Ohio. Responsible person, Robert C. Hendricks, organization code 5300, (216) 433-7507.

12a. DISTRIBUTION/AVAILABILITY STATEMENT

Unclassified - Unlimited
Subject Categories 16 and 99 37

12b. DISTRIBUTION CODE

13. ABSTRACT *(Maximum 200 words)*

Seals Workshop of 1993 code releases include SPIRALI—for spiral grooved cylindrical and face seal configurations; IFACE—for face seals with pockets, steps, tapers, turbulence, and cavitation; GFACE—for gas face seals with "lift pad" configurations; and SCISEAL—a CFD code for research and design of seals—cylindrical configuration. GUI (graphical user interface) and code usage was discussed with hands on usage of the codes, discussions, comparisons, and industry feedback. Other highlights for the Seals Workshop-93 include environmental and customer driven seal requirements; "what's coming"; and brush seal developments including flow visualization, numerical analysis, bench testing, T-700 engine testing, tribological pairing and ceramic configurations, and cryogenic and hot gas facility brush seal results. Also discussed are seals for hypersonic engines, dynamic results for spiral groove and smooth annular seals, and a three-wave journal bearing. Military engine goals include double thrust/weight, decrease SFC 40 percent, decrease secondary air flow leakage 60 percent, increase mainshaft speed 50 percent ; cited tests include 40 seals tested to 1200 °F, 1080 fps, 60 psid, and 0.0045 inch rotor runout with excursions to 0.019 inch for seals to 20 inches in diameter. Future activities will move toward face, lip, proprietary seals, and dynamics.

14. SUBJECT TERMS

Seals; Numerical; Code; Flow

15. NUMBER OF PAGES

393

16. PRICE CODE

A17

17. SECURITY CLASSIFICATION OF REPORT	18. SECURITY CLASSIFICATION OF THIS PAGE	19. SECURITY CLASSIFICATION OF ABSTRACT	20. LIMITATION OF ABSTRACT
Unclassified	Unclassified	Unclassified	